USAAVLABS TECHNICAL REPORT 66-89

DRAG MEASUREMENTS IN A STREAM WITH SINUSOIDALLY FLUCTUATING VELOCITY IN THE LONGITUDINAL DIRECTION

By

Clifton G. Wrestler, Jr.

November 1966

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

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ERRATA

to

USAAVLABS Technical Report 66-89

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FLUCTUATING VELOCITY IN THE LONGITUDINAL DIRECTION

Page 12. Change fourth from last sentence to read as follows:

"The data presented in Figure 5 for Models A, B, and C are plotted in Figure 6 along with the Davenport data.[6]"

Page 13. Change the labels above the curves as indicated:

"Model: Disk" should read "Model A"
"Model: CY I" should read "Model B"
"Model: CY II" should read "Model C"
"Model: SQ I" should read "Model D"
"Model: RP I" should read "Model E"
"Model: RP II" should read "Model F"
"Model: RP III" should read "Model G"
"Model: SQ II" should read "Model H"

ABSTRACT

The drag coefficients of cylinders and plates in a stream with sinusoidal fluctuations in velocity are investigated. A tube with counterrotating vanes at the rear is used to produce the flow fluctuations. This tube was placed in the open jet of a low speed tunnel. The models were mounted on a vertical sting, which supported the one-component force transducer. The drag coefficients, based on average values of drag and dynamic pressure, show no variation with frequency parameter changes. The fluctuating component of drag does show a dependence on the frequency parameter. Also, a phase shift is noticed between the velocity and the force.

FOREWORD

The work reported herein was accomplished while the author was in attendance at the von Karman Institute, a NATO postgraduate school for fluid dynamics.

The author wishes to thank Mr. P. E. Colin and Mr. J. Smolderen of the von Karman Institute for their advice and counsel and the staff of the Low Speed Tunnel for their assistance in coordinating the work in the shops and in performance of the tests.

CONTENTS

ILLUSTRATIONS

SYMBOLS

A	Volume of air represented by model dimensions
B	Coefficient of mass
C_d	Drag coefficient
d	Equivalent diameter of frontal area
D	Drag force
f	Frequency (cycles/sec)
k	Reduced frequency (fd/U)
P	Pressure
q	Dynamic pressure
s	Surface of integration
S	Frontal area of models
t	Time
V	Velocity, instantaneous
U	Average velocity
u	Amplitude of velocity fluctuations
ρ	Density of air
ω	Frequency (rad/sec)
ϕ	Phase angle

SUBSCRIPTS

A	Average quantities
o	Quantity based on U
s	Steady-state values
u	Unsteady quantities

INTRODUCTION

The effect of turbulence in wind tunnel measurements has been recognized and investigated for many years. Some of the early work was performed at the National Bureau of Standards in the United States.[1] Currently, an extension of this work is under way at the National Physical Laboratory in Great Britain. The early tests were performed to explain the effect of turbulence on the drag of flat plates. This work was performed at low turbulence levels, since it is desired to have a very low turbulence level in wind tunnels.[2,3] The investigation presently under way in Great Britain is aimed at developing ways of simulating atmospheric turbulence in wind tunnels. One of the techniques used to obtain high levels of turbulence is to place a grid in front of the test section.[4,5]

Associated with atmospheric turbulence are gusts. Gusts are random fluctuations of velocity, partly caused by the separation of the boundary layer of the earth. It is fortunate that mathematical tools, such as the Power Spectral Analysis, are available to permit the use of harmonic oscillations in testing and to relate the results from these tests to results due to random fluctuations. Experiments are much easier to set up and analyze if harmonic motion is considered.

The experiments discussed herein were devised to explore the effect of a stream with velocity fluctuations on the drag force. Davenport[6] shows a large variation of the drag coefficient with reduced frequency, but very little data are given as to the test conditions and range of parameters. Furthermore, the definition of drag coefficient is not clear. The Davenport model was allowed to oscillate as a pendulum in a uniform stream while the drag force was measured. In the experiments described in the report, a fluctuating stream and a fixed model were used. The extreme condition of the experiment would be one where the stream is oscillating completely with the model fixed; this is not possible with the present flow oscillator. McNown[7] and Keulegan and Carpenter[8] used a hydrodynamic tank to reach the condition of a fully oscillating flow.

To determine the effect of a velocity with sinusoidal fluctuations on the drag force, it is desirable to use models with little or no Reynolds number dependence. First, the disks and plates similar to the standard wind tunnel calibration models were tested. Then, in order to explore the possible effect of the depth of the model on the unsteady drag, models

with the same frontal area, but with various depths, were tested.

This is a basic research program aimed at providing basic information. There are many applications for unsteady drag data; for example, in the case of the drag of helicopter blades in forward flight where large velocity fluctuations occur or in the case of a structure that is buffeted by the winds of a hurricane.

THEORETICAL CONSIDERATIONS

The theory for three-dimensional models, such as those tested, is not available in the literature. However, the theoretical work done on oscillating two-dimensional flat plates normal to the stream is presented in reference 9, where the Helmholtz flow model is used. Similar work with two-dimensional models in a sinusoidal stream has been presented by McNown[7] and investigated further by Keulegan and Carpenter.[8] In reference 8, the force is given in three parts:

$$D = A\rho \frac{d}{dt}(BV) + \oint P ds + \frac{1}{2}\rho C_d S V^2. \tag{1}$$

Equation 1 is derived from the equation of motion by integrating and applying Green's theorem. This integration is done in reference 8 for two-dimensional flow with the assumption of irrotational and incompressible flow. Careful examination of each of the three terms in equation (1) shows that the first term is due to acceleration and the last term to velocity. The middle term is a pressure term in the test region without the model. If the area of integration is large enough, the term P is nothing more than the hydrostatic pressure. If air is used as a test medium, it can be shown that the integral around the surface of integration is negligible.

The velocity can be represented as a steady-state component and as an oscillating component:

$$V = U + u \sin(\omega t). \tag{2}$$

The time derivative of V yields

$$\frac{dV}{dt} = \omega u \cos(\omega t). \tag{3}$$

Substituting equations (2) and (3) in equation (1) gives an equation that can be expressed as a steady-state part plus three fluctuating components:

$$D = \frac{1}{2}\rho C_d S U^2 + \frac{1}{2}\rho C_d S \left[2Uu \sin(\omega t)\right] + \frac{1}{2}\rho C_d S u^2 \sin^2(\omega t) + AB\rho\omega u \cos(\omega t). \tag{4}$$

Hence,

$$D = \frac{1}{2}\rho C_d S U^2 \left[1 + \frac{2u}{U}\sin(\omega t) + \left(\frac{u}{U}\right)^2 \sin^2(\omega t)\right] \tag{5}$$

$$+ AB\rho\omega u\cos(\omega t)$$

or

$$D = D_o\left[1 + \frac{2u}{U}\sin(\omega t) + \left(\frac{u}{U}\right)^2 \sin^2(\omega t)\right] + AB\rho\omega u\cos(\omega t). \tag{6}$$

The average dynamic pressure is obtained by integrating the velocity squared over 1 cycle.

$$q_A = \frac{\rho\omega}{4\pi}\int_o^{\frac{2\pi}{\omega}}\left[U^2 + 2Uu\sin(\omega t) + u^2\sin^2(\omega t)\right] dt, \tag{7}$$

which yields

$$q_A = \frac{1}{2}\rho\left(U^2 + \frac{1}{2}u^2\right) . \tag{8}$$

The average drag is the mean value of the measured drag force. The drag coefficient can be expressed in terms of average drag and average dynamic pressure:

$$C_d = \frac{D_A}{Sq_A} . \tag{9}$$

The approximation that

$$q_A \simeq \frac{1}{2}\rho\ U^2 = q_o$$

yields an error that is less than one-half of 1 percent for values of fluctuating velocity that are less than 10 percent of the average velocity. The drag coefficient can be expressed as

$$C_d \simeq \frac{D_A}{Sq_o} . \tag{10}$$

With the same approximation, the force can be expressed as

$$D = D_o + D_u \sin(\omega t + \phi), \tag{11}$$

where

$$D_o = \frac{1}{2}\rho C_d S U^2 = q_o C_d S$$

and

$$D_u = \frac{1}{2}\rho \left[(C_d S 2 U_u)^2 + (2AB\omega u)^2\right]^{\frac{1}{2}} .$$

To determine the magnitude of the out-of-phase component of drag, assume that it is zero. The drag equation (11) for sin (ωt) of unity (there is nothing to cause a phase shift) becomes

$$D = D_o + D_u = D_o \left(1 + \frac{2u}{U}\right) . \tag{12}$$

Dividing through by D_o yields

$$1 + \frac{D_u}{D_o} = 1 + \frac{2u}{U} \tag{13}$$

or

$$\frac{\frac{D_u}{D_o}}{\frac{2u}{U}} = 1. \tag{14}$$

If in reality the out-of-phase component does not exist, the results from equation (14) will yield a value of 1 at all times.

Another approach to determine the out-of-phase component is to measure the phase shift. The phase angle can be obtained mathematically from equation (11):

$$\phi = \text{ARCTAN}\left(\frac{AB\omega}{UC_dS}\right) . \tag{15}$$

Upon examination of the terms within parentheses, it is discovered that the phase angle is a function of only reduced frequency for a given model:

$$\phi = \mathcal{F}(k) . \tag{16}$$

EXPERIMENTAL INVESTIGATION

APPARATUS

The dimensions and the relative position of the model and vanes are given in Figure 1. The flow through the flow oscillator, which is shown in Figure 2, is obtained by placing the oscillator in the open jet of a low speed wind tunnel (see reference 10). The vanes, which are 120mm wide by 955mm long, at the rear of the tube counterrotated to avoid disturbing the fluid normal to the flow direction. Because of the boundary conditions, the flow was not quite symmetric over one revolution of the shaft. This asymmetry was less than 1 percent, based on the maximum dynamic pressure obtained during a static cycle. This fluctuation was not desirable, but a complete redesign of the flow oscillator would have been required to correct it (see appendix).

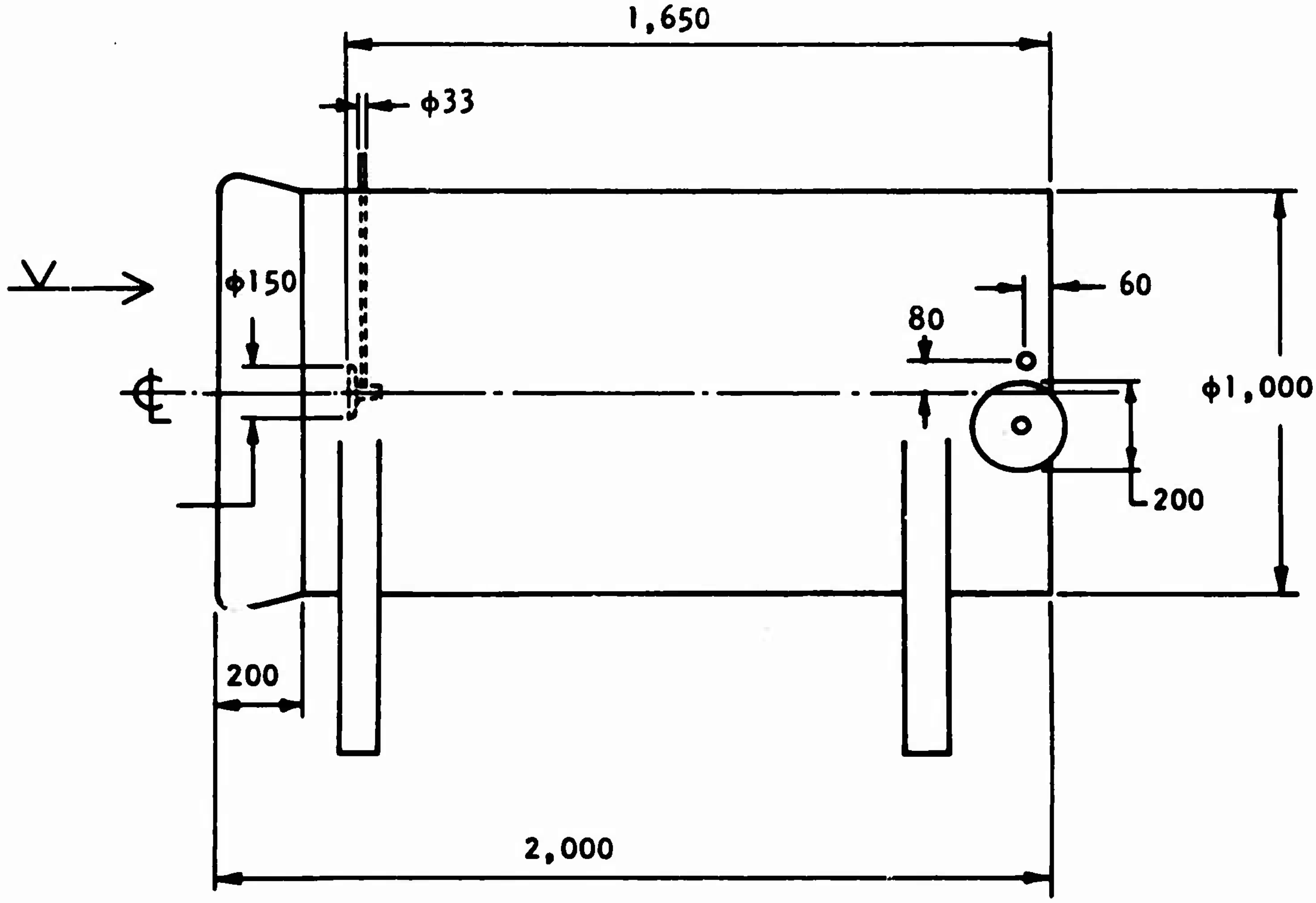
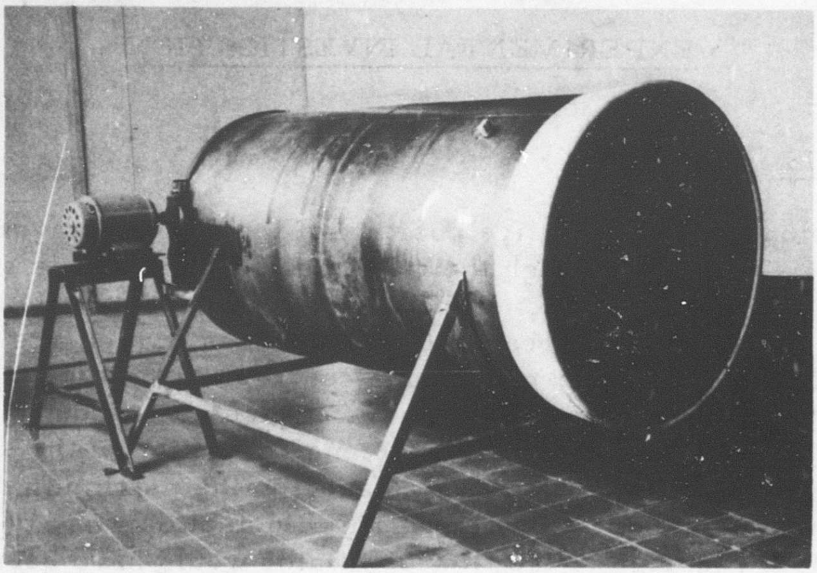
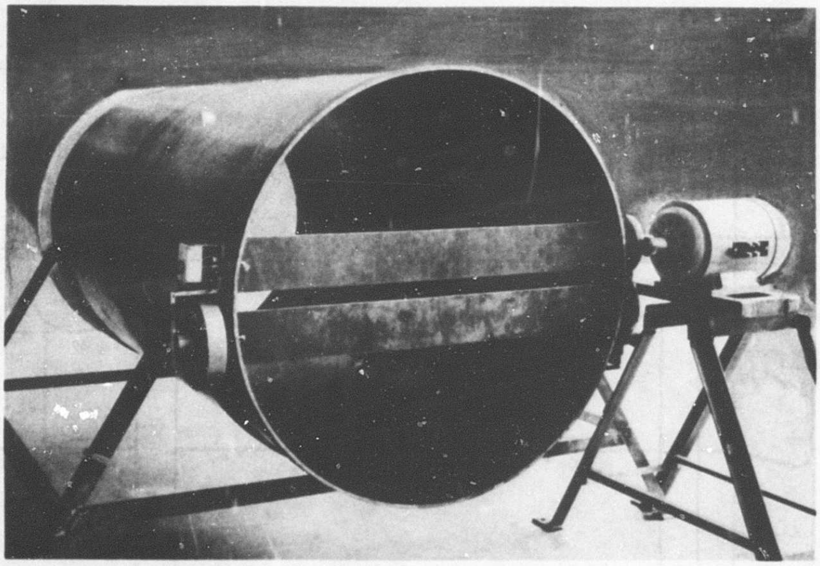


Figure 1. Sketch of Flow Oscillator. (Dimensions are in millimeters.)



Right Front View



Left Rear View

Figure 2. Photographs of Flow Oscillator.

MODELS

The models were sting-mounted from the existing balance platform. A dash-pot was installed to reduce the vibration level at the balance to a minimum.

The circular, square, and flat-plate models and their dimensions and weights are shown in Figure 3. The maximum and minimum diameters are shown for the circular models. The circular models were not turned on a lathe; therefore, they were slightly out of round. The equivalent diameter based on the frontal area is given in meters, since this is what the reduced frequency is based on. The actual frontal area is also tabulated.

INSTRUMENTATION

The instrumentation used for the tests was standard equipment that is used with static and dynamic measurements. The wiring diagrams for the three active elements are shown in Figure 4. Unfortunately, the necessary electronic equipment for using the resolver as a computing resolver was not available; therefore, the resolver was used only as a position indicator for the vanes. The hot wire and the model were installed at the same station on the flow oscillator. The hot wire[14] was connected in a constant current bridge, and the current through the five micron wires was adjusted to 80 milliamperes. The outputs from the balance, hot wire, and resolver were recorded simultaneously on a CEC recording oscillograph. A Statham transducer with a full-scale range of ± 30 grams and with a linearity of 0.15 percent was used. Since the signals from the hot wire and from the balance were not pure sine waves, the reading of phase shift and of amplitude of fluctuation was difficult. Following is a complete list of equipment:

Hot Wire

Power Supply: 0- to 15-volt DC, Type BED 001
Milliammeter: Boite de Controle, Type 4885, No. 472044
Decade Box: Heathkit, Model IN-11

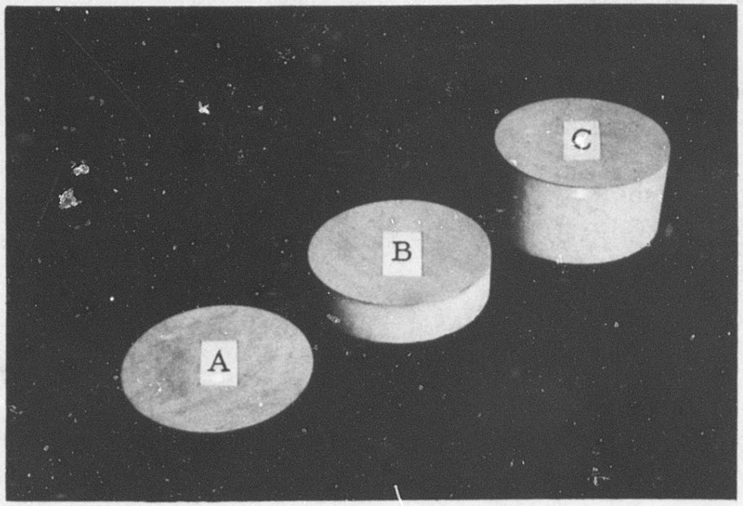
Balance

Power Supply: 0- to 15-volt DC, Phillips, DE4803
Transducer: Statham, ±30 grams
Filter: Series-type inductance-capacitance, 50 cps

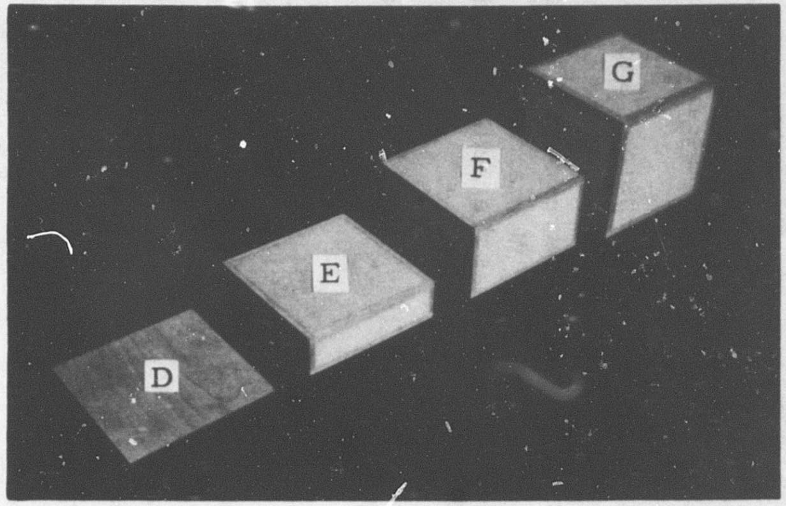
Resolver

Power Supply: Heathkit Audio Generator, Model AG-9
Resolver: Clifton Precision Product Co., Type 9

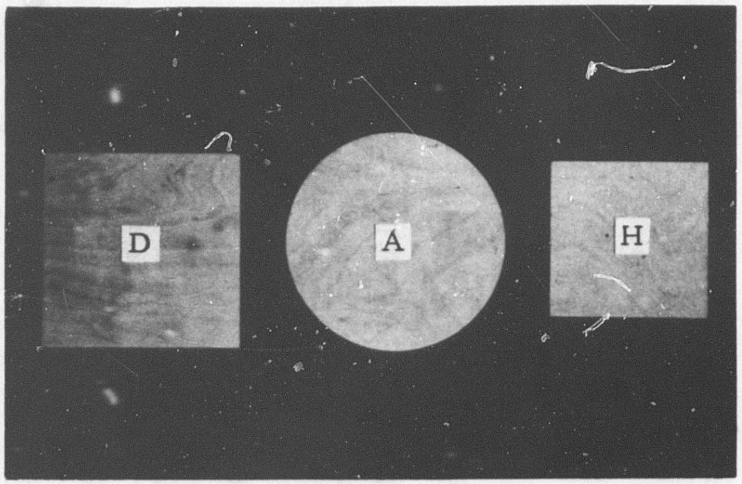
Circular Models



Square Models



Flat-Plate Models



MODEL DIMENSIONS AND WEIGHTS

Model	Length Along Axis of Flow (mm)	Length, Horizontal (mm)	Length, Vertical (mm)	Weight (gm)	Diameter (m)	Frontal Area (m^2)
A		149.5	149.5	39.38	0.15	0.01754
B	81.2	150.5	150.0	46.96	0.15	0.01772
C	40.7	150.9	150.6	41.24	0.15	0.01784
D		133.7	133.5	40.71	0.15	0.01785
E	132.6	132.5	131.6	87.44	0.15	0.01742
F	79.7	133.4	132.8	36.40	0.15	0.01771
G	39.9	132.7	132.3	24.01	0.15	0.01753
H		106.3	106.3	26.78	0.12	0.01131

Figure 3. Models With Dimensions and Weights.

Event Marker

Power Supply: Hot wire power supply
Decade Box: Heathkit, Model IN-11

Recorder

Recording Oscillograph: Consolidated Electronics Corp., Model 5-124

Velocity Standard

Velometer: Alnor, Type 3002, Serial No. 9878

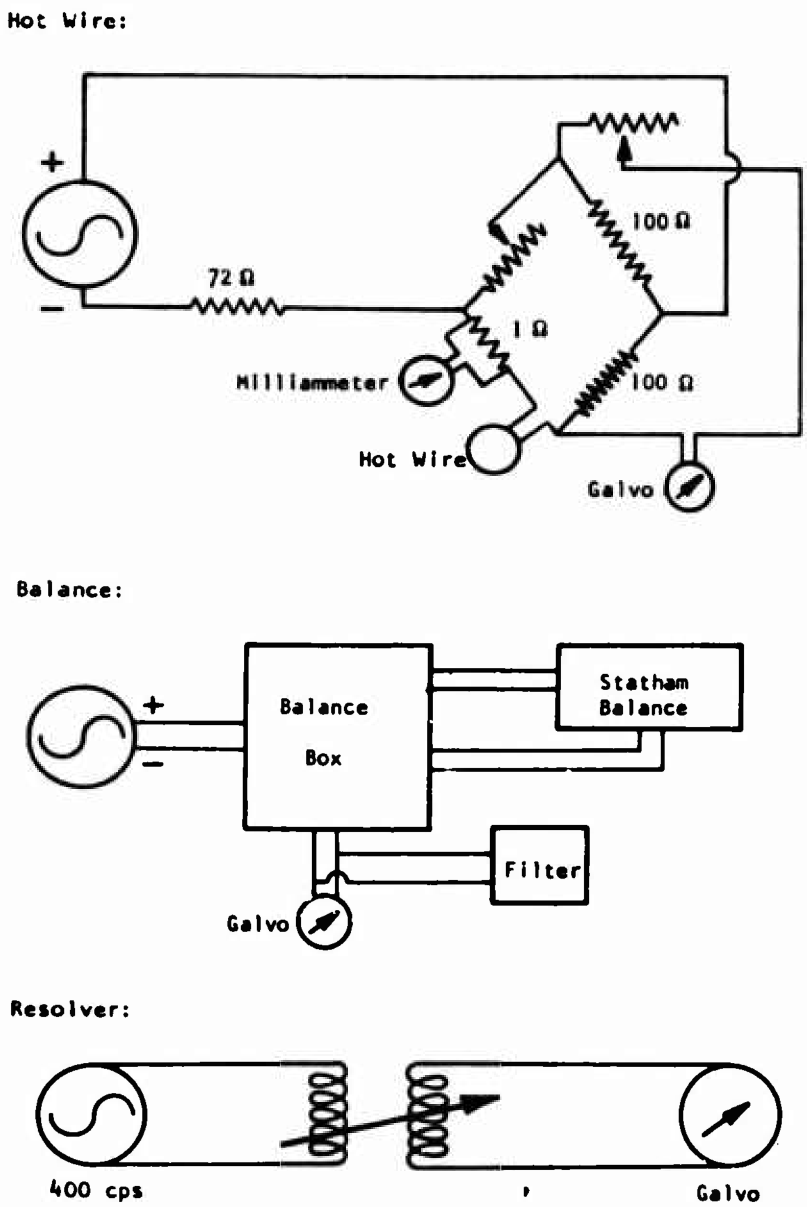


Figure 4. Wiring Diagrams of Instrumentation.

PROCEDURES AND RESULTS

TESTING TECHNIQUE

Prior to each sweep in frequency, the tunnel speed was adjusted to the desired value. The hot wire and the load on the balance were recorded at various static velocities by introducing blockage by means of the vanes at the rear of the flow oscillator. The balance sensitivity was adjusted for the range of dynamic pressures encountered during the tests. The velocity was measured with a velometer having known characteristics. This gave a calibration to use with each record. This calibration was also checked at the end of each sweep in frequency. With the calibration finished, the velometer was removed from the flow oscillator; the testing was then performed by increasing the frequency from about 0.8 cps to 3.5 cps and by taking a record of the balance and hot wire outputs at various discrete frequencies.

DRAG MEASUREMENTS

The investigation was intended to show that the drag coefficient increased with reduced frequency, as shown in reference 6. The results from this investigation, as shown in Figure 5, are based on average drag and the average dynamic pressure; see equation (10). The drag coefficient is normalized by the steady-state drag coefficient to emphasize the trend with reduced frequency. The values for the steady-state drag for disks, plates, and cylinders can be found in Hoerner.[11] The values of drag coefficient of plates are virtually independent of Reynolds number over the range of test speeds used. Further information on disks can be found in Knight[12] and Shoemaker.[13] However, as pointed out in reference 1, the drag values vary with the amount of turbulence. Therefore, it is best to use the calibration records taken before each run. Figure 5 shows no variation of the drag coefficient, as defined by equation (10), with a reduced frequency of up to 0.25. Some of the tests were extended to a reduced frequency of 0.6, and there was still no effect of reduced frequency on the drag coefficient. The data presented in Figure 5 for the disk, CY I, and CY II models are plotted in Figure 6 along with the Davenport data.[6] The data from reference 6 is normalized with respect to the value given for steady-state drag coefficient. The drag coefficient does not vary, as is indicated; further analysis of the data was made in order to understand the pheonomena involved and to explain the discrepancy between the two techniques used to measure the unsteady drag. Figure 7 shows a typical recording of the data.

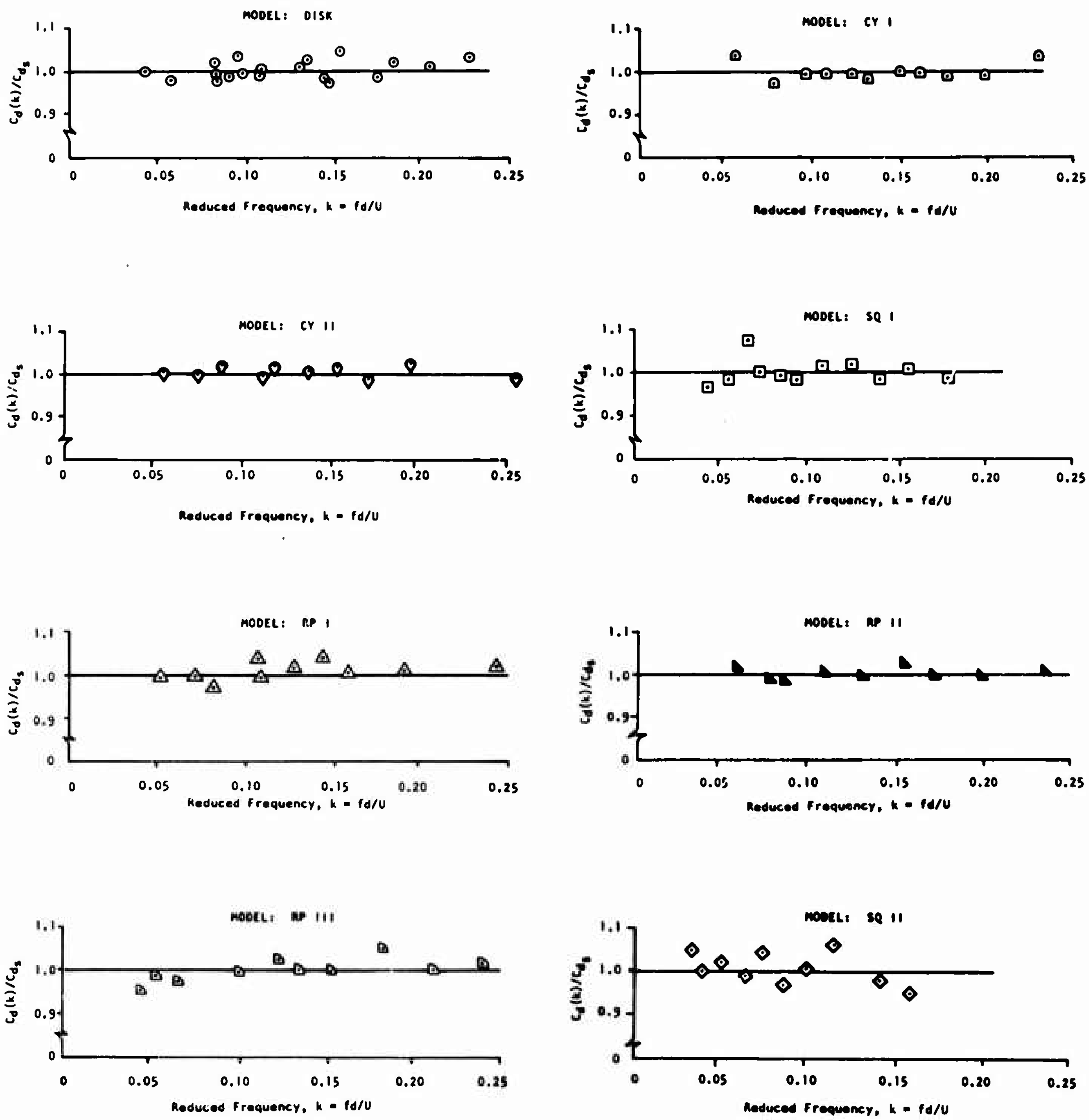


Figure 5. Normalized Drag Coefficient as a Function of Reduced Frequency.

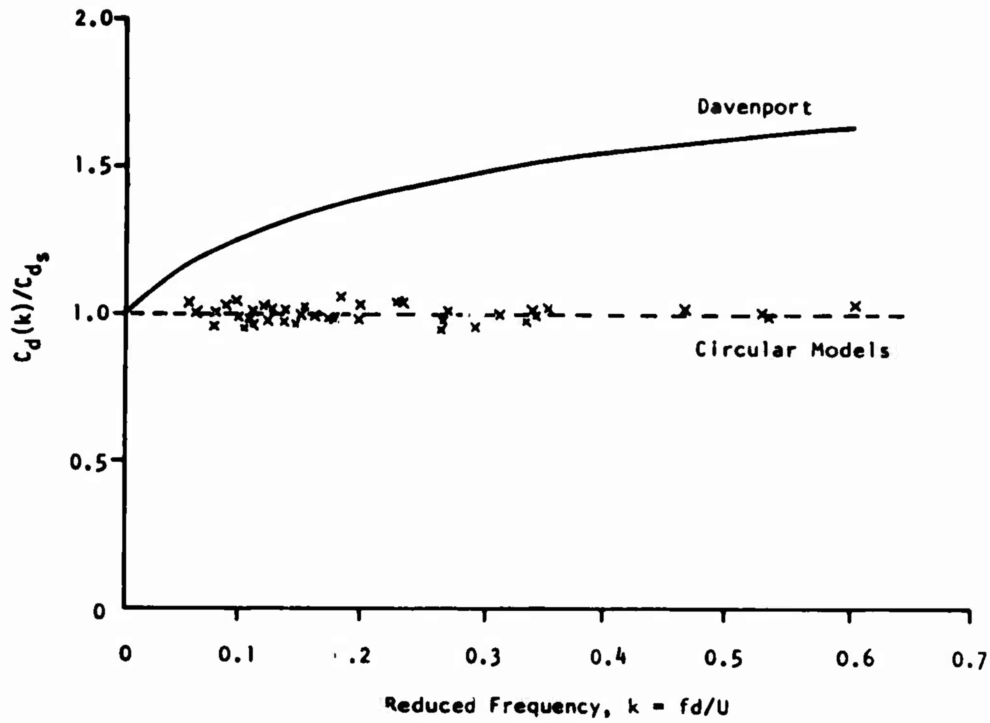


Figure 6. Comparison of Drag of the Circular Models With Data From Davenport.

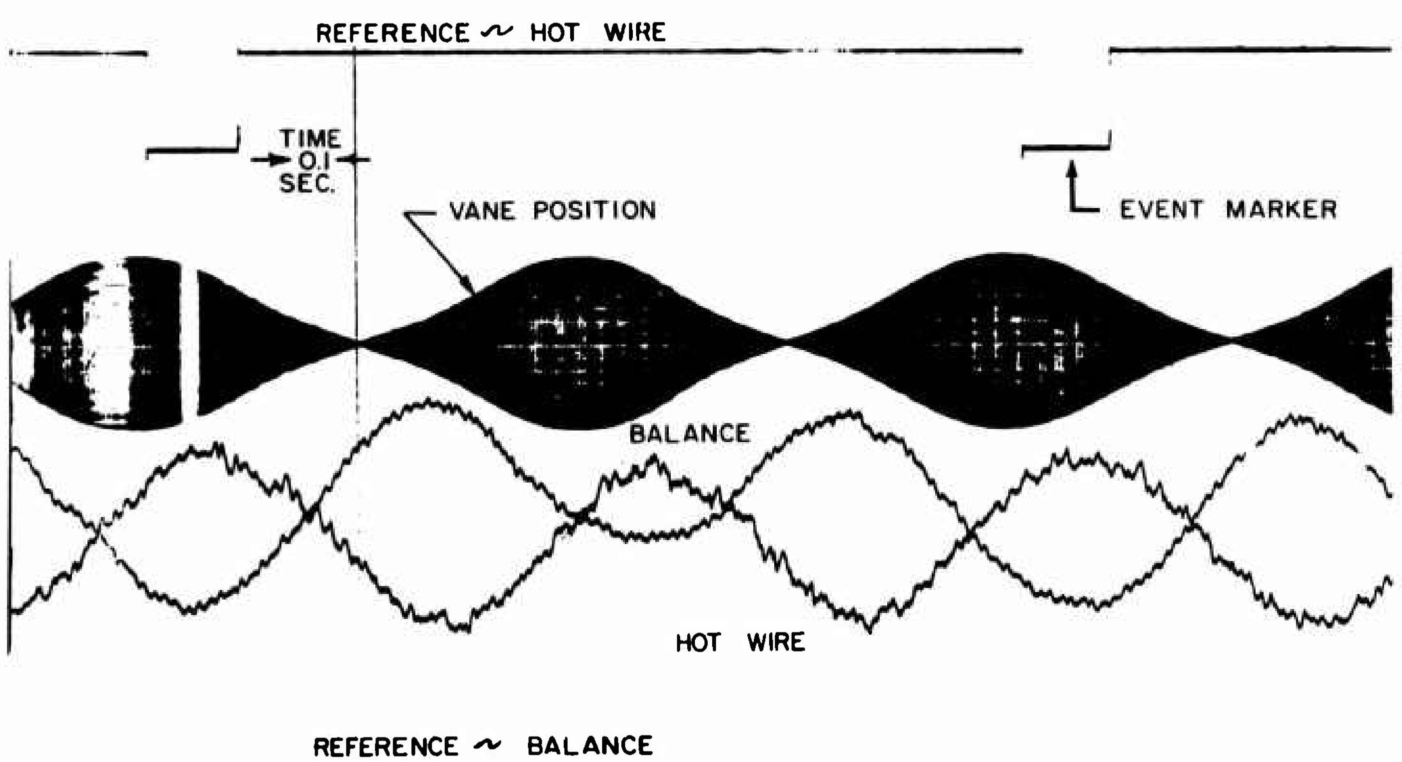


Figure 7. Typical Data Record.

UNSTEADINESS EFFECTS

If the unsteady portions of the record and the theoretical discussion following equation (11) are considered, it can be shown that the left-hand portion of equation (15) is not 1. Furthermore, the results presented in Figure 8 show that incremental drag is an increasing function with increasing reduced frequency. Figure 9 shows the approximate phase shift between the velocity fluctuations and the force fluctuations. The word *approximate* should be emphasized, since the method of reading phase difference is not very accurate. Another factor contributing to the difficulty in making the unsteady measurements is shown in Figure 10. The ratio of amplitude of the velocity fluctuations to the average velocity decreases quite rapidly up to a reduced frequency of 0.1, based on the model diameter, and more slowly up to a reduced frequency of 0.25. The limiting case of a reduced frequency of zero yields a value of $\frac{u}{U}$ of 0.26 for the speeds used; however, as the reduced frequency is increased to 0.1, the value of $\frac{u}{U}$ is decreased to 0.032.

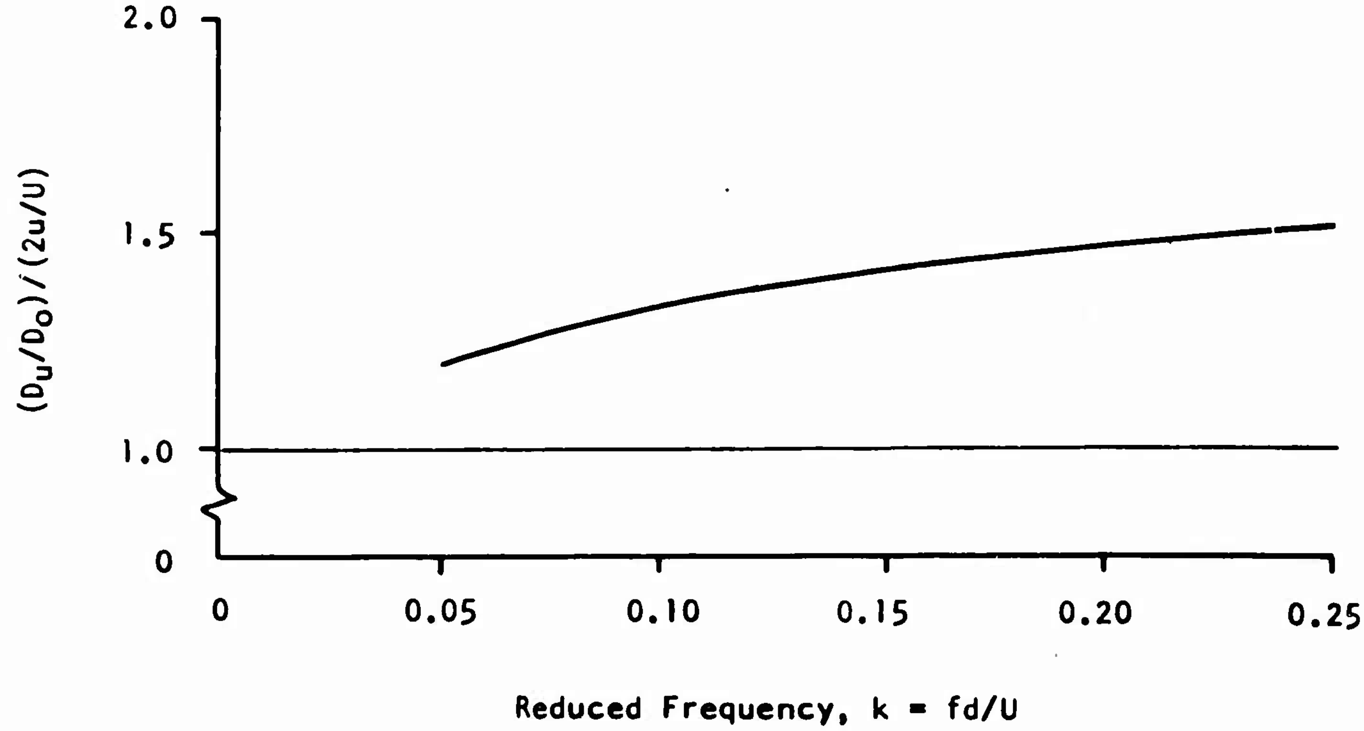


Figure 8. Incremental Drag as a Function of Reduced Frequency; Disk Model.

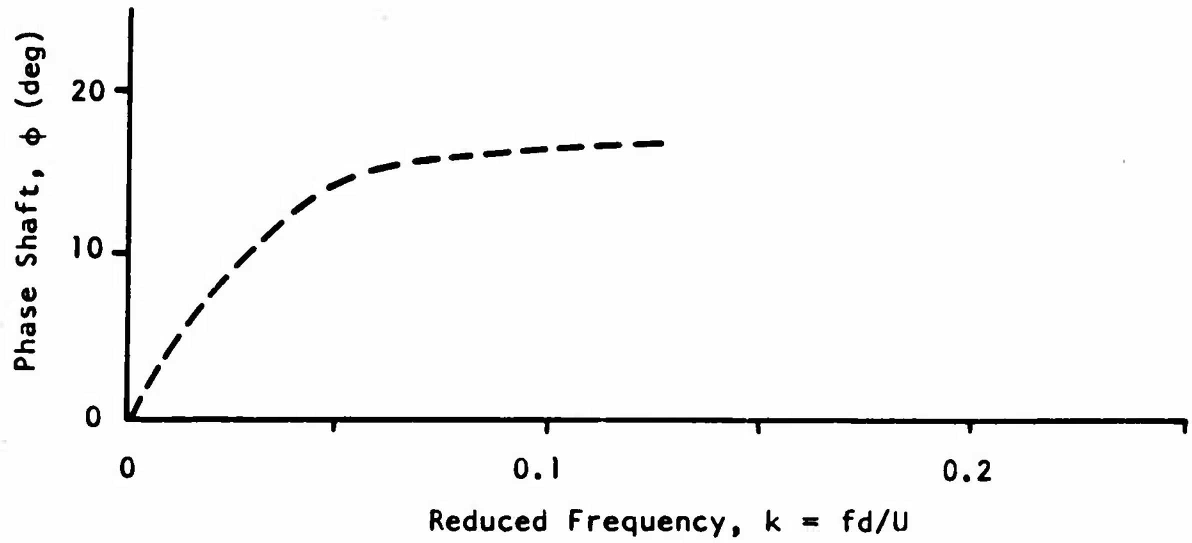


Figure 9. Phase Shift in Unsteady Drag Versus Reduced Frequency; Disk Model.

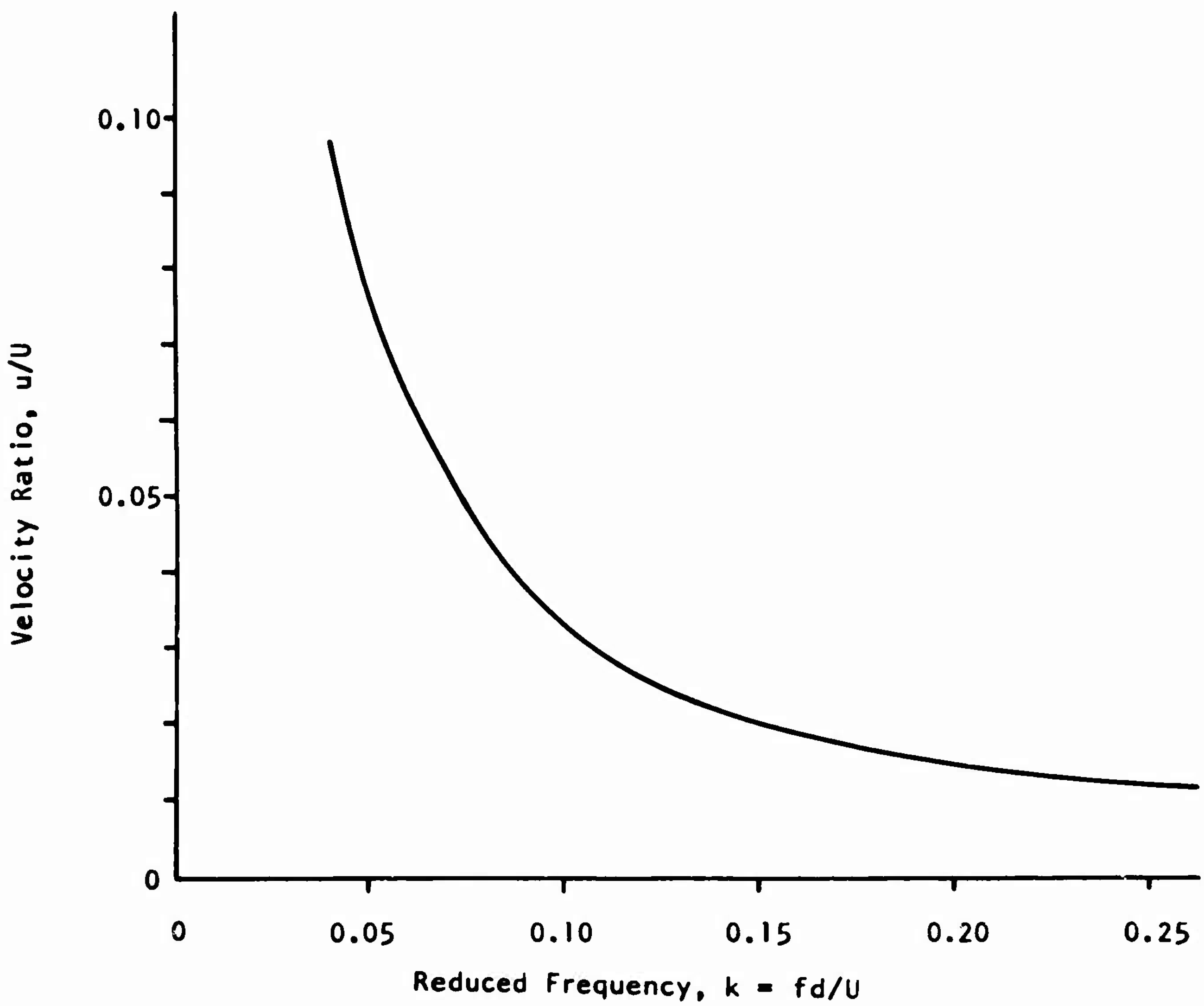


Figure 10. Amplitude Ratio of Unsteady to Average Velocity With Reduced Frequency; Disk Model.

CONCLUSIONS

The drag coefficients based on average values do not show a frequency dependence. However, there is a component of the force which is in quadrature and is frequency-dependent. This investigation did not have the proper instrumentation to investigate the quadrature component fully, but results were obtained from the records taken. Poor accuracy is all that could be expected for the phase angle measurements between the velocity at the model and the force. It would be better to measure the velocity and the force through either an electromechanical or an electronic resolver system.

EXTENSION OF WORK IN UNSTEADY FLOW

If the investigation discussed in this report is continued, the following ideas would be worthy of consideration:

1. A tube with a square or rectangular cross-sectional area could be built as a flow oscillator, with a large ratio of blade area to cross-sectional area. The overall length of the flow oscillator could, and should, be reduced for the same size models. The model would be closer to the vanes, but it should not be too close because of the possible high local velocities produced at the vanes. With such an arrangement, four or six blades could be used.

2. Instrumentation should include a resolver system with two input channels so that the relative phase between the velocity and force could be measured. For example, a system could have a two-stage mechanical computing resolver (such as that made by Clifton Precision Products Company, Type MPR 23A1) and appropriate electronic equipment. This resolver would also give the information concerning the operation of the vanes. For the hot wire, a linearizer is almost essential to produce a signal that could be measured with greater ease when the ratio of fluctuating velocity to average velocity is very small.

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APPENDIX

FLOW OSCILLATOR

DESIGN

The original flow oscillator was designed as a tube with a diameter of 100 centimeters and a length of 200 centimeters. The leading edge was faired with an annular-shaped airfoil. The axis of the tube was aligned with the flow direction. The flow modulator was a cone with a 110-centimeter diameter and a 90-degree apex angle. This cone was not satisfactory, since it produced too much blockage. A 40-centimeter-diameter cone was tried next. This cone produced the proper amount of blockage; however, to produce the proper variation, a 60- 80-centimeter displacement was required. This was not considered to be desirable.

Different flat plates were placed at the rear of the flow oscillator, and the drop of dynamic pressure was recorded. The results agree well with the continuity equation, with the assumption that the flow through the section where the flat plates are placed is the free-stream velocity. The results are shown in Figure 11. With this information, the present system was designed; rotating flat plates were used to produce the flow fluctuations. Two vanes were used; they rotated in opposite directions about an axis passing through what would be considered as the midchord. The spacing of the vanes was governed by the availability of matched gears. This nonoptimum configuration produced an irregularity in the cycle: it departed from a sine wave for over half a turn of the drive shaft. The results are plotted in Figure 12.

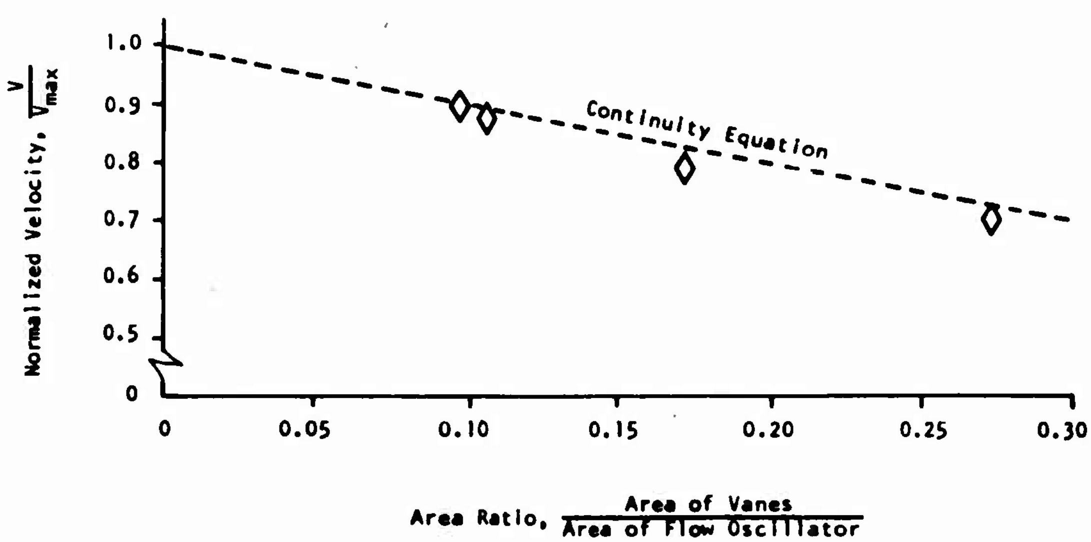

Figure 11. Variation of Velocity in the Flow Oscillator as a Function of Blockage Area Ratio.

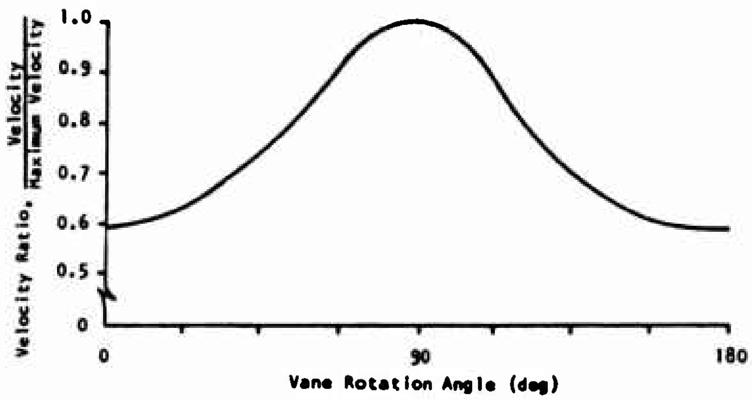

Figure 12. Velocity Variation Over One Static Cycle.

PERFORMANCE

The present configuration was designed to produce a 25-percent variation in velocity. This variation was achieved statically only. The frequency dependence of the amplitude of fluctuation was quite large. This variation of fluctuating velocity is shown in Figure 13. The same falloff in the amplitude would be expected for a different relative vane area, but the magnitude would be different. The phase angle between the velocity and the position of the vanes is given in Figure 14. To a first approximation, the phase angle would be only a function of reduced frequency for a given geometry of the oscillator.

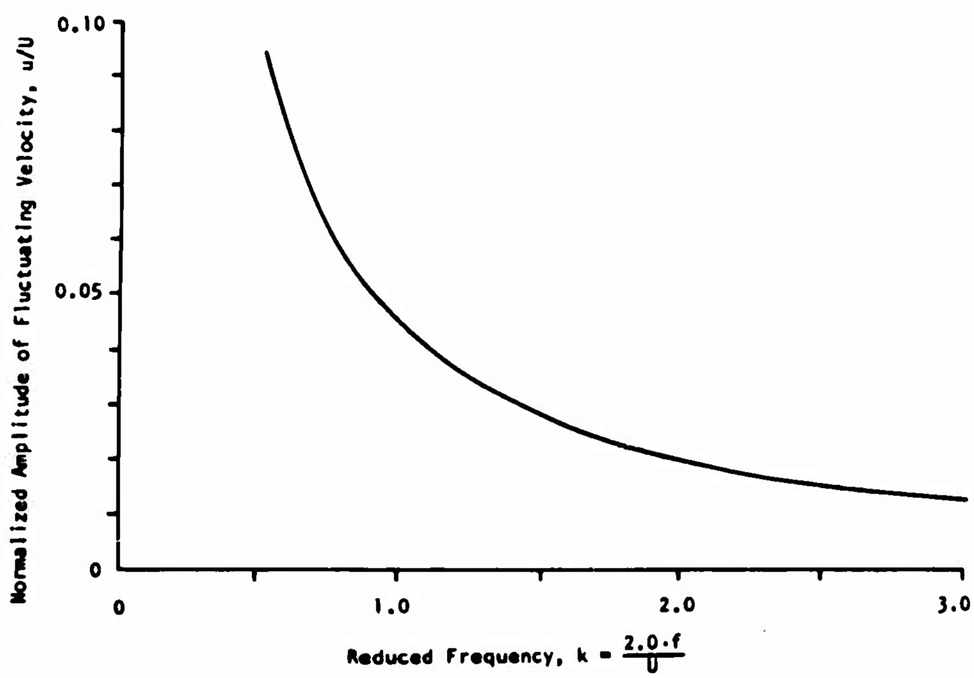

Figure 13. Variation of Amplitude of Fluctuating Velocity With Reduced Frequency.

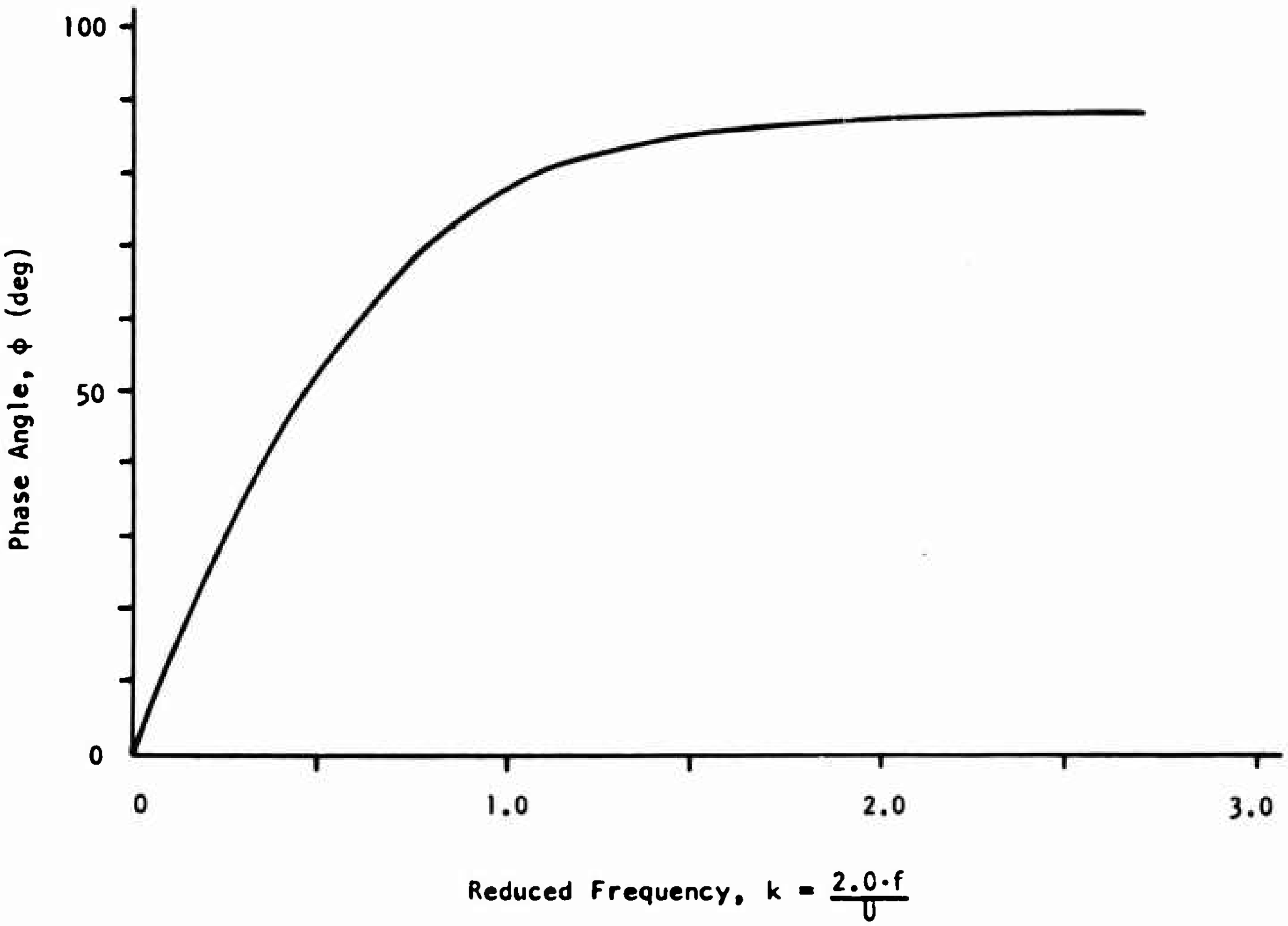

Figure 14. Measured Phase Angle at 159 Centimeters Ahead of the Vanes.

PROPOSED SYSTEM FOR FUTURE TESTS

If a similar flow oscillator is used in the future, the following design suggestions would be worthy of consideration. A tube of square cross section, 1 meter in length and 1 meter on each side, could be built. The leading edge could be faired by a 15-percent Clark Y airfoil with about a 30-centimeter chord. The axis of the vanes could be located 15 centimeters forward of the rear edge of the tube. Careful attention should be devoted to the spacing of the vanes to avoid the problem encountered during the cycle in the present flow oscillator. The drive system should be well designed and balanced to reduce vibration levels and to minimize friction. Six vanes should be used in the oscillator and driven as shown in Figure 15. The vane dimensions are not given in Figure 15, since it is proposed that the total vane area be explored by using three areas: 0.5, 0.7, and 0.9 square meter.

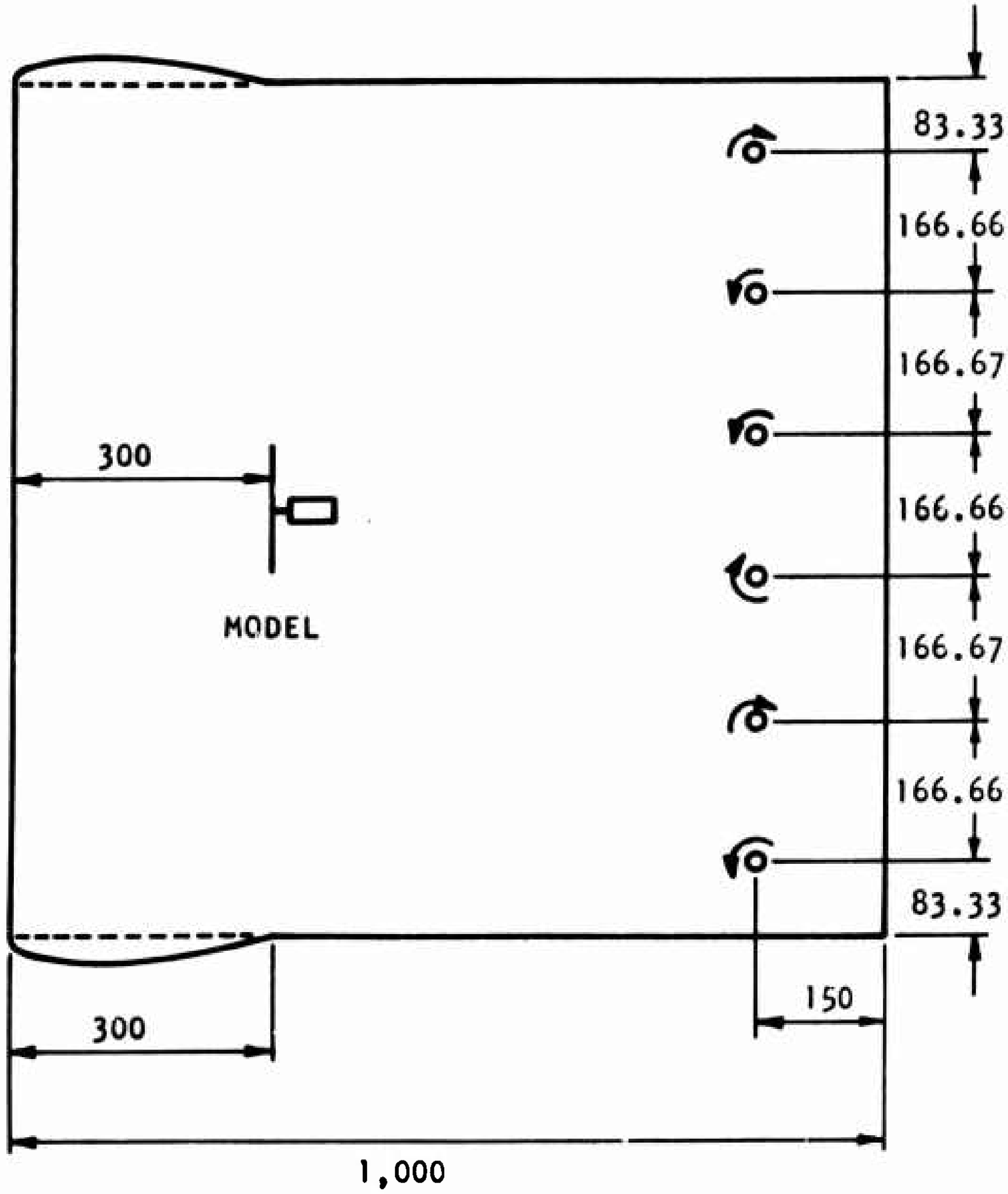

Figure 15. Side View of Proposed Oscillator. (Measurements are in millimeters.)

PROPOSED INSTRUMENTATION FOR FUTURE TESTS

The instrumentation for future tests could be similar to that used during the exploratory tests with the following changes: A linearizer should be used with the hot wire to obtain the output proportional to the velocity fluctuations. Various types of mechanical and electronic resolver systems comparable to the Type 23 resolver made by Clifton Precision Products Company should be used. In order to use the resolver properly, an oscillator with a modulator, a demodulator, and an integrator is needed. A galvanometer with a large damping factor could be used as an integrating device.

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13. ABSTRACT

The drag coefficients of cylinders and plates in a stream with sinusoidal fluctuations in velocity are investigated. A tube with counterrotating vanes at the rear is used to produce the flow fluctuations. This tube was placed in the open jet of a low speed tunnel. The models were mounted on a vertical sting, which supported the one-component force transducer. The drag coefficients, based on average values of drag and dynamic pressure, show no variation with frequency parameter changes. The fluctuating component of drag does show a dependence on the frequency parameter. Also, a phase shift is noticed between the velocity and the force.

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